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Lubrication

A Technical Publication Devoted to the Selection and Use of Lubricants

This Issue

Trends in Lubrication of Large Low-Speed Diesels

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Trends in Lubrication of Large Low-Speed Diesels

DIESEL ENGINES or compression ignition engines use liquid fuels of low volatility varying from kerosine and distillate, to crude oil. Compression ratios vary from 11.5:1 to 22:1 and compression pressures from 400 to 700 pounds per square inch.

These engines are made in a variety of cylinder sizes and number, cylinder arrangements, and valve arrangements. Cylinder arrangements may be vertical, horizontal opposed, double opposed, Vee, radial, or angled with pumps or compressor pistons driven from a common crankshaft. Either two or four-stroke cycle operation is used and the engine may be single or double acting. Intake and exhaust control is effected by poppet valves, sleeve valves, cylinder ports or combinations of these.

The simplified sketches in Figure 1 show the operation of the four-stroke diesel cycle which uses one separate stroke of the piston for each of the processes of suction (intake), compression, expansion (power) and exhaust. Figure 2 presents similar sketches of the two-stroke diesel cycle which eliminates separate suction and exhaust strokes by performing these functions at the end of the expansion or power stroke.

The four-stroke cycle as compared to the two-stroke permits higher piston speeds, wider variations in speed and load, cooler pistons, higher mechanical efficiency, lower specific fuel consumption and lower pumping losses. The two-stroke cycle shows 50 to 80 per cent greater power output per cylinder for the same displacement and speed, twice as many power strokes per cycle, piston and crankpin pressures in one direction up to speeds at which piston inertia force exceeds compression gas pressures on the pistons, and the lower cost permitted by a valveless engine.

Large low speed diesels are commonly constructed with the bottom end of each cylinder open to the crankcase. In this design the lower portions of the cylinder liner are wetted with crankcase lubricant thrown from the crankshaft. Oil is also supplied to the piston and cylinder liner through openings in the cylinder walls from external metering pumps. One characteristic of this design is that lubricant on the cylinder walls is contaminated with products of fuel combustion which find their way past the piston rings and piston skirt into the crankcase to contaminate the crankcase oil, particularly if the fuel in use is a heavy fuel that is not completely burned.

The *trunk piston* design is shown in Figure 3. Figure 4 illustrates another engine design referred to as the *crosshead* type which seals the crankcase from the piston around the piston rod. In this design

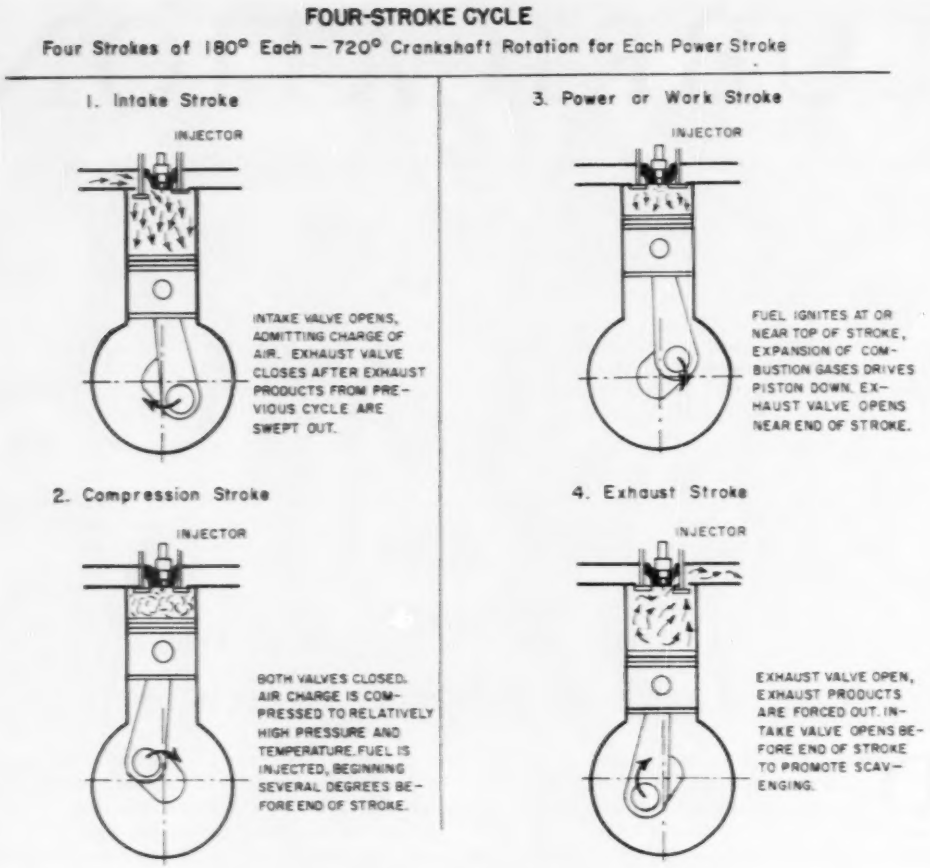

FOUR-STROKE CYCLE
Four Strokes of 180° Each — 720° Crankshaft Rotation for Each Power Stroke
1. Intake Stroke
INJECTOR
INTAKE VALVE OPENS, ADMITTING CHARGE OF AIR. EXHAUST VALVE CLOSES AFTER EXHAUST PRODUCTS FROM PREVIOUS CYCLE ARE SWEPT OUT.
2. Compression Stroke
INJECTOR
BOTH VALVES CLOSED. AIR CHARGE IS COMPRESSED TO RELATIVELY HIGH PRESSURE AND TEMPERATURE. FUEL IS INJECTED, BEGINNING SEVERAL DEGREES BEFORE END OF STROKE.
3. Power or Work Stroke
INJECTOR
FUEL IGNITES AT OR NEAR TOP OF STROKE, EXPANSION OF COMBUSTION GASES DRIVES PISTON DOWN. EXHAUST VALVE OPENS NEAR END OF STROKE.
4. Exhaust Stroke
INJECTOR
EXHAUST VALVE OPEN, EXHAUST PRODUCTS ARE FORCED OUT. INTAKE VALVE OPENS BEFORE END OF STROKE TO PROMOTE SCAVENGING.

Figure 1 — Four-stroke Cycle Diesel.

the underside of the piston is frequently used as a compressor piston to supply scavenge air to the intake manifold. Figure 5 illustrates three types of two-cycle diesel engines.

In many modern diesel engines the exhaust gases are passed through a turbine wheel which is connected directly to a centrifugal type air compressor which supplies air under pressure to the intake manifold. This method of using exhaust gas energy to supply higher pressure air to the engine intake system is known as *turbocharging*. Another method employs gear-driven or reciprocating compressors to supply higher pressure air and is known as *supercharging*.

The power produced by one cylinder of a diesel engine is expressed in the following equation.

$$BHP = \frac{PLAN}{33000}$$

where: BHP = Brake Horsepower

P = BMEP (brake mean effective pressure on the piston head) in pounds per square inch

L = Length of power stroke in feet

A = Area of piston in square inches

N = Number of power strokes/minute

From the above formula is it obvious that to increase the power output of a cylinder of fixed stroke and bore (cylinder diameter) it is necessary to increase either the "P" (BMEP) or the "N" (number of power strokes per minute). Increasing either the quantity of air or fuel to the cylinder may increase the BMEP, however increasing fuel alone leads to over-rich combustion, smoking, and to higher combustion temperature which can exceed

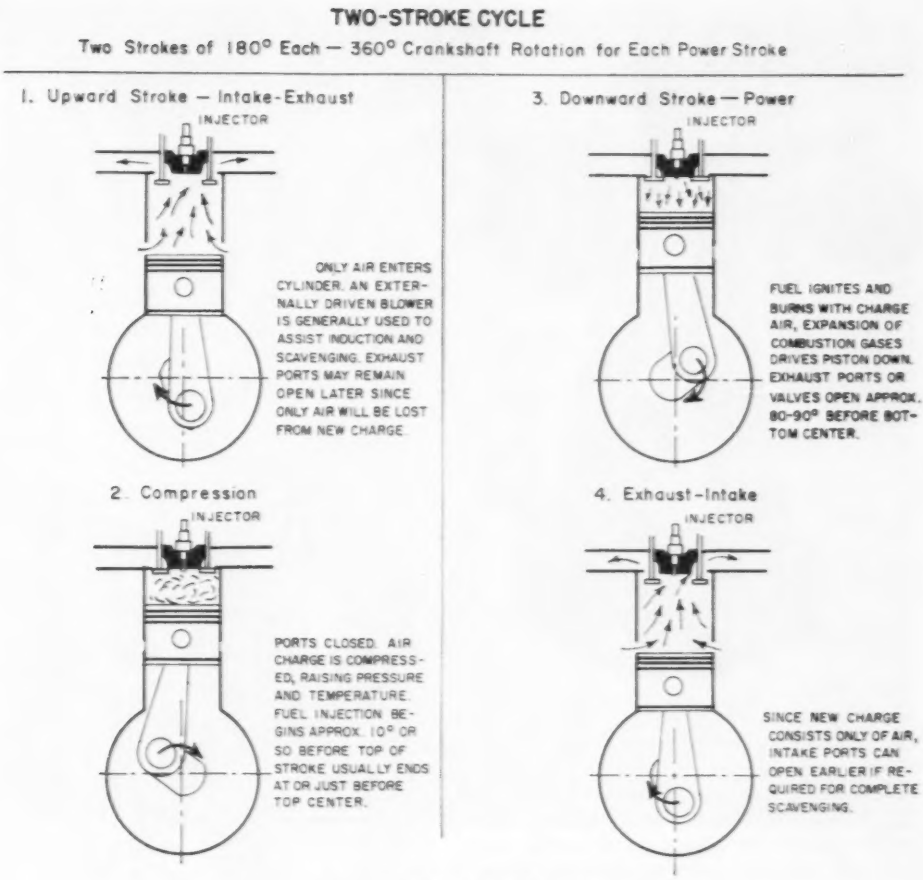

TWO-STROKE CYCLE
Two Strokes of 180° Each — 360° Crankshaft Rotation for Each Power Stroke
1. Upward Stroke — Intake-Exhaust
INJECTOR
ONLY AIR ENTERS CYLINDER. AN EXTERNALLY DRIVEN BLOWER IS GENERALLY USED TO ASSIST INDUCTION AND SCAVENGING. EXHAUST PORTS MAY REMAIN OPEN LATER SINCE ONLY AIR WILL BE LOST FROM NEW CHARGE.
2. Compression
INJECTOR
PORTS CLOSED. AIR CHARGE IS COMPRESSED, RAISING PRESSURE AND TEMPERATURE. FUEL INJECTION BEGINS APPROX. 10° OR SO BEFORE TOP OF STROKE USUALLY ENDS AT OR JUST BEFORE TOP CENTER.
3. Downward Stroke — Power
INJECTOR
FUEL IGNITES AND BURNS WITH CHARGE AIR, EXPANSION OF COMBUSTION GASES DRIVES PISTON DOWN. EXHAUST PORTS OR VALVES OPEN APPROX. 80-90° BEFORE BOTTOM CENTER.
4. Exhaust-Intake
INJECTOR
SINCE NEW CHARGE CONSISTS ONLY OF AIR, INTAKE PORTS CAN OPEN EARLIER IF REQUIRED FOR COMPLETE SCAVENGING.

Figure 2 — Two-stroke Cycle Diesel.

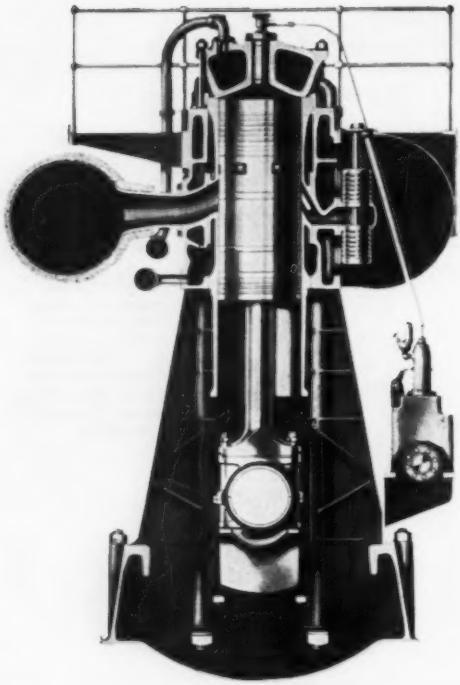
the safe limit of operation for the engine components. To gain maximum BMEP *both* the quantity of air and fuel to the cylinder must be increased in the correct proportion to maintain safe combustion and exhaust temperatures. Supercharging or turbocharging is therefore used to force more air into the cylinder which can then be used to burn more fuel. The obvious manner of increasing the number of power strokes per minute in a particular design engine is to increase its rotational speed, which is limited by such elements of design as the strength of parts, permissible bearing loads, destructive vibration, etc.

DIESEL PROGRESS

The diesel engine has undergone continuous changes since the latter part of the 19th Century when Rudolph Diesel patented his "rational heat engine". These changes often resulted from major engineering development but much more frequently from refinements designed to increase the efficiency of the cycle and improve its reliability in service. Design of units for a wide range of specific applications has made the diesel engine a prime contender as a power source over a broad field of power ranges.

Large low speed diesel engines like those illustrated in Figures 6, 7, 8 and 9 have shared a large part in these developments. The major trend in the last ten years has been toward increase in horsepower per cylinder through increases in cylinder pressures and higher rotational speeds. Although some increase in cylinder presures has been gained through better engine breathing, turbocharging has had the greatest effect.

Turbocharging of four-cycle diesel engines has been practiced for many years. Today, turbocharging is being made available in two-cycle engines and from the many advantages gained, it is expected this



Courtesy of Nordberg Manufacturng Co.

Figure 3 — Trunk-piston type diesel.

development will be adopted at an accelerated pace.

The increase in horsepower available per cylinder has made possible the production of engines with considerably increased power ratings with little increase in either size or weight. The year 1957 witnessed the construction of the first 15,000 BHP single acting two-stroke engine and many more of this power are on order. Plans are already advanced by several engine builders for the manufacture of 20,000 HP and larger units. With thermal efficiencies of 39% for these diesel units compared with 25 per cent for steam turbine marine power and with diesel engines with these power ratings becoming available for ships up to 100,000 tons in displacement, the diesel engine is becoming an increasingly attractive source of marine power.

In marine terminology a vessel powered by a diesel engine is called a *motor ship*. Steamships employ either steam turbines or reciprocating steam engines.

The course of motor ship and diesel engine output for the years 1953 through 1957 is shown in Table I. The year 1938 is shown for comparison. The increasing use of diesel power is well apparent.

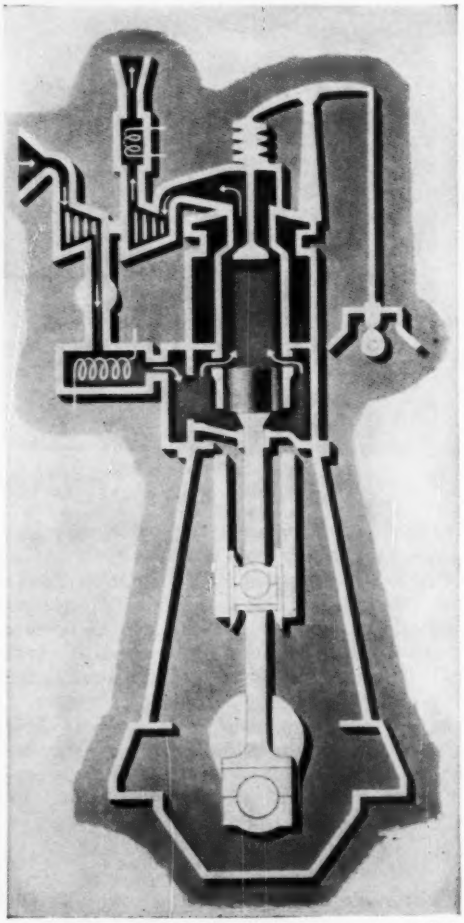
Motor Ships represented 25 per cent of the total

TABLE I

Motor Shipbuilding Output
(Above 2,000 Deadweight Tons)

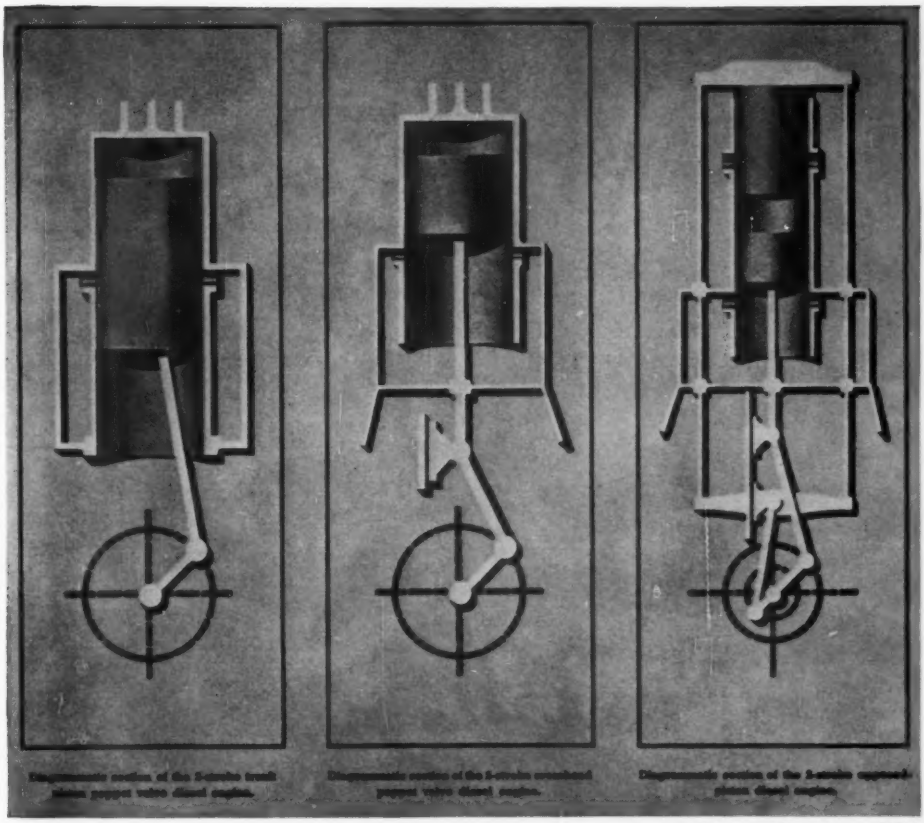
World

Year	Motor Ship Tons Dwt.	Machinery BHP	Tons/HP
1938	2,200,000	1,020,000	2.16
1953	3,810,000	2,130,000	1.79
1954	3,671,000	1,198,000	3.06
1955	3,743,000	2,156,000	1.74
1956	4,330,000	2,450,000	1.77
1957	5,969,000	3,250,000	1.84



Courtesy of Burmeister and Wain

Figure 4 — Schematic of cross-head type four-cycle diesel.



Courtesy of Burmeister and Wain

Figure 5 – Schematics of three types of two-cycle diesels.

marine vessel tonnage in 1950. It is estimated that the diesel will power 40 percent of the world's tonnage by the end of this year. In 1957 ship construction, diesel power was used in 60% of the world's tonnage output.

FUTURE OUTLOOK

Looking toward tomorrow's diesel engines, it is not expected that they will differ markedly in *basic* construction from those of today. They will undoubtedly reflect many design refinements along the following lines:

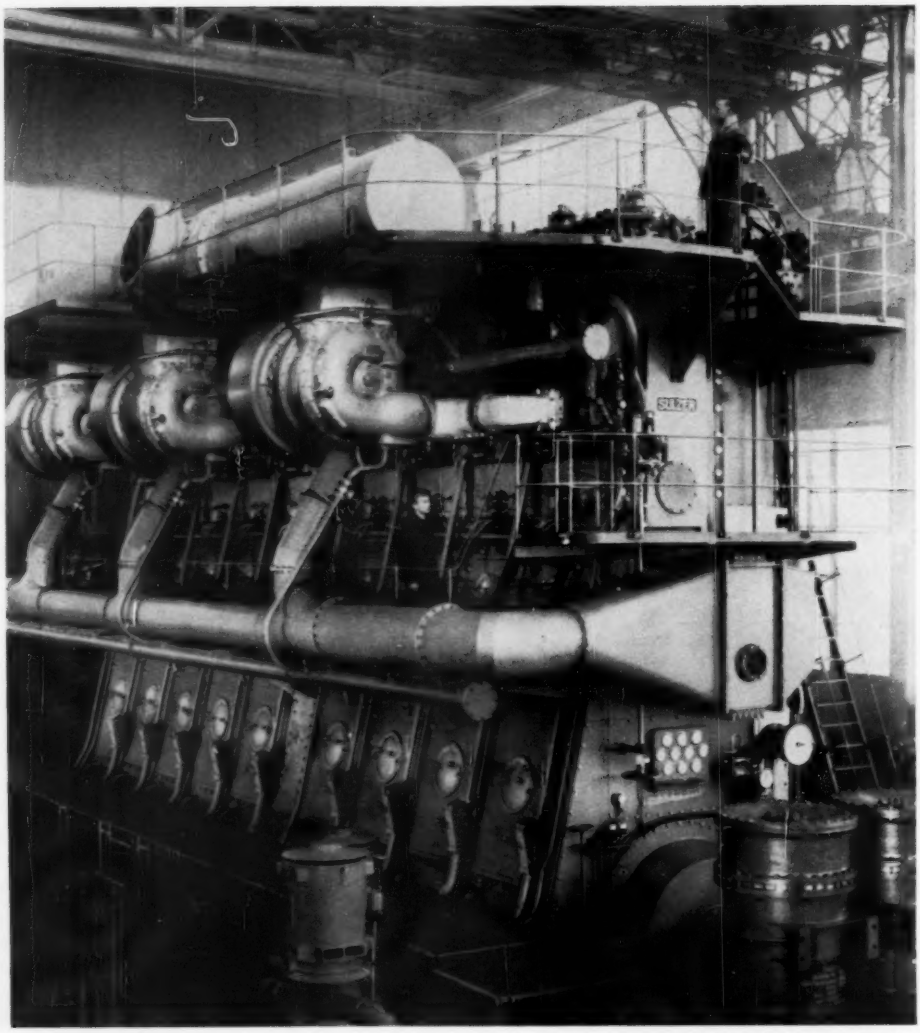
Higher power and efficiency

Higher rotational speeds. Rotational speeds have increased slowly in the past and this trend is expected to continue. Resistance to higher engine and piston speeds is not as great today and is not considered as important an index of engine performance as in the past.

Improved air breathing capacity. Refinements to reduce restrictions in the air supply to the engine and the exhaust system will improve the efficiency of moving air through the engine. Design of systems to take advantage of air pulsation will be increasingly used.

Turbocharging will be applied to an increasing extent on both two-cycle and four-cycle engines. Turbocharging pressures will likewise be increased. Figure 10 is an estimate of the brake mean effective pressures of two and four-cycle engines for the next 10 years which will result directly from this increased turbocharging pressure. Combining turbocharging with intercoolers will increase the thermal efficiency of the engine.

Improved combustion. Intensive research continues on all phases of combustion. Results obtained in improved performance by adaptation of surface ignition principles make this area of research and development active. Improved methods and effi-



Courtesy of Sulzer Brothers, Ltd.

Figure 6 — 11,700 BHP single-acting nine-cylinder two-cycle marine engine with exhaust-gas turbo chargers.

ciencies in the application of supercharging and turbocharging are expected to yield improved thermal efficiencies.

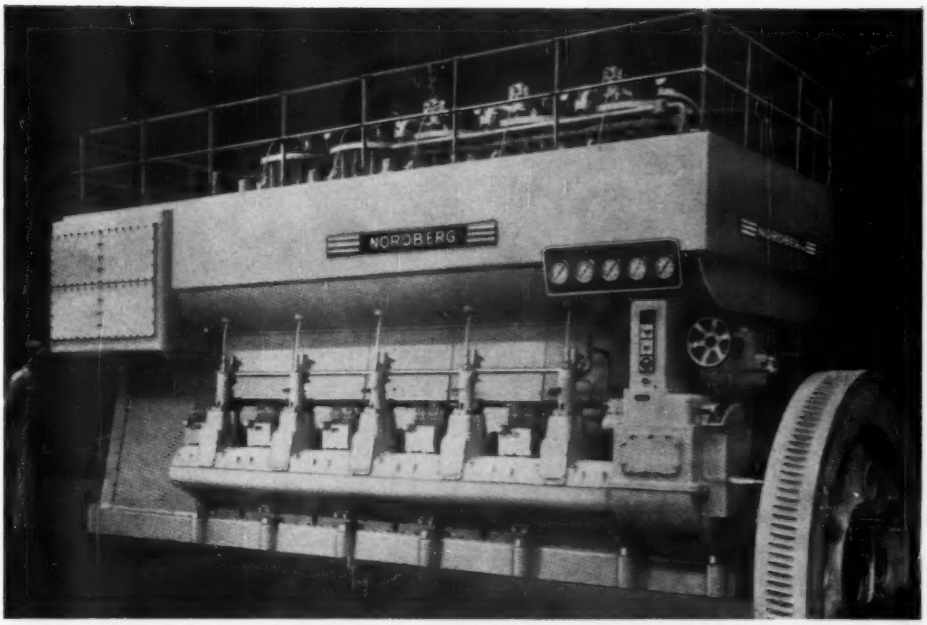
Higher BMEP. More power from a given size of engine maintains its space requirement while reducing cost of installation.

Improved Reliability and Maintenance

Better noise control. This problem undoubtedly will become increasingly difficult as more stringent limitations are applied. Research and development on silencing methods can be expected to provide many answers.

Better designs and materials. New and better materials with improved service performance are continuously being made available for bearings, pistons, liners, rings and other engine parts. These materials will find increasing application and new materials will be made available.

Better maintenance. Simplification and better accessibility of parts will promote easier and better maintenance.



Courtesy of Nordberg Manufacturng Co.

Figure 7 — 2150 BHP single-acting five-cylinder two-cycle blower-scavenged engine.

FUELS

For many years, the fuel appetite of diesel engines was satisfied with distillate type fuels. With these, and improvements in design and simplification, diesel operation has become increasingly reliable and trouble-free.

Confident of the diesel's dependability and with the goal of improved economy, diesel operators have recently been using heavy "residual" type boiler fuels or blends of these with distillates. These fuels cost less per gallon and contain more heat energy or potential power per gallon. With the experience gained in developing the required handling procedures for boiler fuels, their use has been found quite feasible in large low speed engines. Table II presents some of the physical characteristics of fuel oils and illustrates that there is some overlapping among the five grades. It should be noted that our primary interest in this article lies in the use of the heaviest or No. 6 or Bunker C grade. Following are descriptions of these grades as defined by ASTM Designation D396:

Fuel Oil — *Definition*

No. 1 — A distillate oil intended for vaporizing pot-type burners and other burners requiring this grade of fuel.

No. 2 — A distillate oil for general purpose domestic heating for use in burners not requiring No. 1 fuel oil.

No. 4 — An oil for burner installations not equipped with preheating facilities.

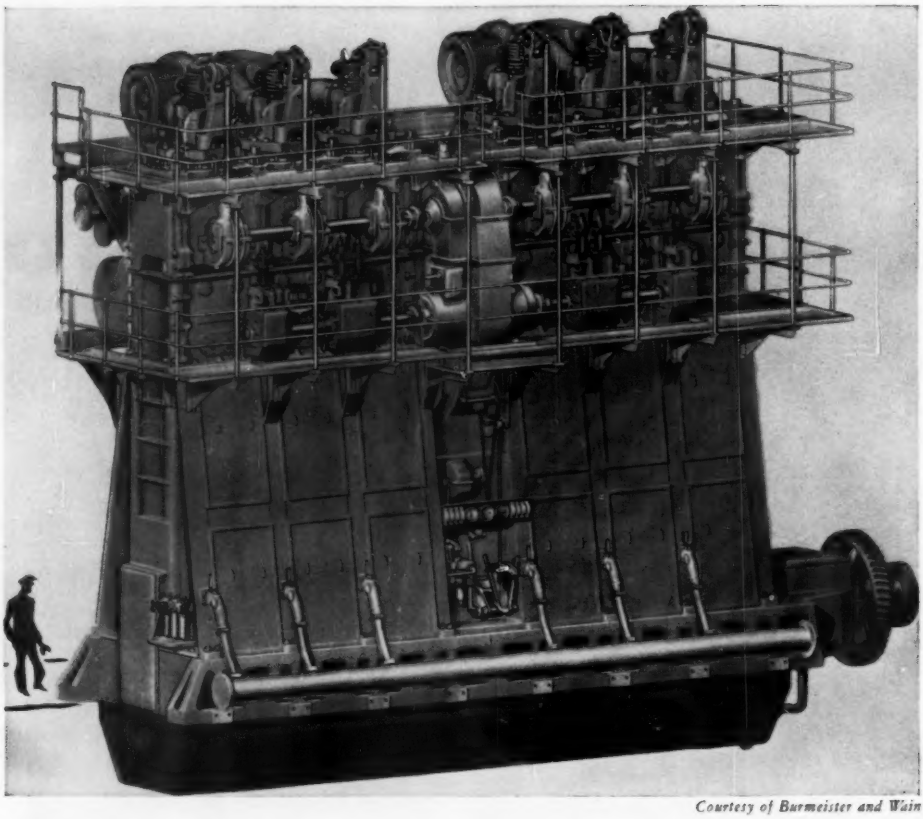
No. 5 — A residual-type oil for burner installations equipped with preheating facilities.

No. 6 — An oil for use in burners equipped with preheaters permitting a high viscosity fuel. (This grade is frequently referred to as Bunker "C" fuel oil.)

UTILIZATION OF HEAVY FUEL

Bunker C fuels contain the major portions of the residual (non-distillable) content of each barrel of crude received by the refinery. Crudes are treated prior to their processing to remove many of the contaminants that would be injurious. However, crude oils contain traces of many metals such as vanadium or sodium and these are concentrated during refining in the heavier oils such as Bunker C.

In handling Bunker C fuel in tankers, through dock lines to terminals, in tank cars, and storage tanks it is frequently subjected to contamination with sea water and fresh water, dust and other



Courtesy of Burmeister and Wain

Figure 8 — 7500 BHP single-acting six-cylinder two-cycle turbo-supercharged engine.

materials. Being viscous and heavy, removal of water and other contaminants is often very difficult, requiring heating, centrifuging and filtering equipment to place the fuel in condition for best use in the diesel engine.

Figure 11 is a schematic diagram of a typical Bunker C fuel processing system used in a stationary or marine diesel installation. For easy pumping the heavy fuel storage tank must be provided with some means of heating the fuel on its way to the pump. To prevent unnecessary deterioration of the fuel, this temperature should be the lowest possible that will still permit good pumping viscosity. For purifying and clarifying, practices often differ. In any case, the fuel must again be heated to facilitate cleaning and adequate flow rates through the centrifuge.

Prior to its arrival at the engine injector pump, the fuel must again be heated. Here it is necessary to maintain a temperature high enough to assure good atomization of the discharge from the injector. Efficient atomization is necessary to assure complete combustion of the fuel. It is best that this heating be done at a point in the fuel system between the header (engine low-pressure fuel line) and the high pressure fuel injector pump. Applying heat in the fuel header or in lines to the fuel header will mean the heating of a large quantity of fuel much of which will be recirculated. Such unnecessary heating of so much fuel in circulation can lead to deterioration of the fuel and cause sludging, deposits and filter problems.

The temperatures necessary in the various stages of the fuel system naturally depend on the characteristics of the fuel, on the particular diesel engine used and on the type of service in which it is engaged.

Figure 12 is a viscosity-temperature plot of fuels of different viscosities. Separate viscosity ranges are shown for easy pumping, centrifuging and injection. The values are typical. The correct value for a specific installation must depend on the recommen-

dations of the engine builder who has established limits through experience.

It is important to know that the viscosity indices of residual fuels vary widely and as a result the temperature to which any one must be heated to attain a certain viscosity will vary. Accordingly, it is a good practice to plot the viscosities of the fuel in actual use to establish the temperature at which it reaches the desired viscosity.

The viscosity of fuel at the injector is a most important consideration. Too high a viscosity will cause incomplete combustion and will lead to exhaust smoke, deposits on valves, deposits in valve ports, and increased contamination of crankcase lubricants. The viscosity necessary varies with engine design, cylinder size and operating conditions.

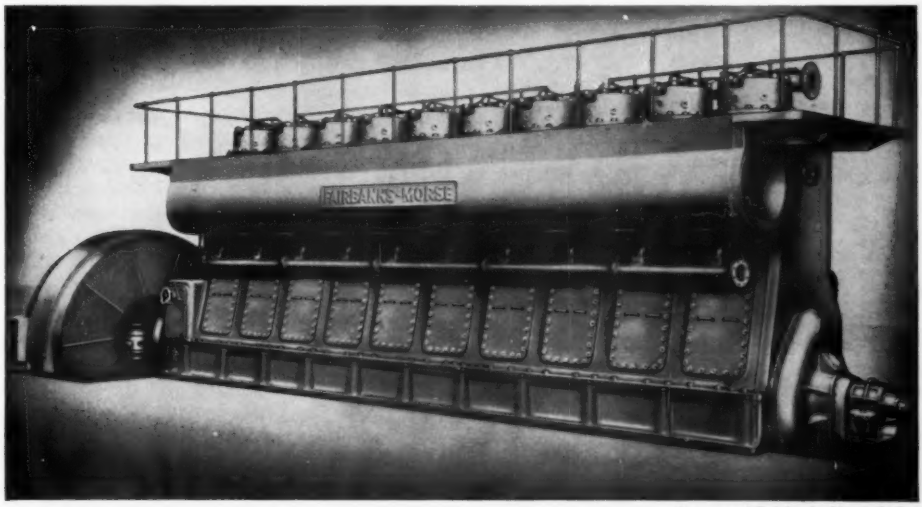
Fuel temperature as well as fuel viscosity affect combustion. Low fuel temperature increases ignition delay (the time for the fuel to start burning in the cylinder after injection) since the colder the fuel the more time is required for it to pick up heat of compression and begin the burning cycle. By the same token, the more heat put into the fuel from outside the engine, the easier it is vaporized, therefore less time will be required for it to pick up the additional amount of heat to start burning.

If the fuel is not atomized nor heated sufficiently, it will not be completely burned in the engine.

Figures 13 and 14 show intake and exhaust ports containing deposits developed when the fuel was not completely burned. These deposits restrict air flow through the engine.

Experience gained through the handling of heavy fuels has resulted in the development of handling practices that have gained the result of satisfactory combustion. It has been learned that even though satisfactory combustion can be accomplished, some problems still remain. Trace metals such as sodium and vanadium, either alone but particularly together, lead to early destructive damage of the poppet type valves through what appears to be a corrosive attack or metal leaching like that shown in Figure 15. Various types of materials have been tried as additives in both fuel and lubricant. The results have been mixed. In some cases real beneficial effect is reported, in others, metal attack continues. The ash from trace metals in some cases aids in formation of deposits on the valve seat causing incomplete seating of the valve and resultant valve burning as illustrated in Figure 16 and highly magnified in Figure 17. Continued research in this field is necessary to gain completely satisfactory results. As heavy fuels vary considerably in their content of vanadium and sodium, selection of fuels from sources with a low content of these materials is at present the best solution. Reduction of sodium content can be accomplished in many cases through water washing. No economic means of removing the vanadium compounds is currently available but research continues.

Another normal characteristic of heavy fuels is their high sulfur content. Many experiments have demonstrated that high wear of cylinder liners (particularly cast iron) and rings will result from the burning of fuels with high sulfur content at normal engine temperatures. During many of the early trials in the use of heavy fuels, the added maintenance



Courtesy of Fairbanks-Morse & Co.

Figure 9 — 3500 BHP single-acting ten-cylinder two-cycle engine.

TABLE II

Physical Measurements of Oil at 60°F.

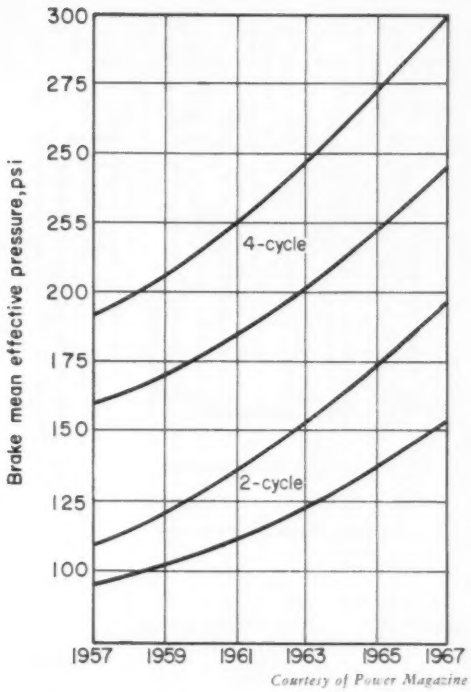
Fuel Oil (CS-12-48) Grade No.		Gravity Degrees A.P.I.	Specific Gravity	Lb. per Gallon	B.t.u. Per Lb.	Gross B.t.u. Per Gal.	Net B.t.u. Per Gal.	Lb. per 42-Gal. Barrel	Weight Lb. per Cu. Ft.
6		3	1.0520	8.76	18,190	159,340	152,100	368.00	65.54
6		4	1.0443	8.69	18,240	158,500	151,300	365.31	65.07
6		5	1.0366	8.63	18,290	157,840	149,400	362.62	64.59
6		6	1.0291	8.57	18,340	157,170	148,800	359.98	64.12
6		7	1.0217	8.50	18,390	156,320	148,100	357.37	63.65
6		8	1.0143	8.44	18,440	155,340	147,500	354.81	63.19
6		9	1.0071	8.39	18,490	155,130	146,900	352.46	62.78
6		10	1.0000	8.33	18,540	154,620	146,200	350.15	62.36
6		11	.9930	8.27	18,590	153,740	145,600	347.71	61.93
6		12	.9861	8.22	18,640	153,220	144,900	345.28	61.50
6	5	14	.9725	8.10	18,740	151,790	143,600	340.53	60.65
6	5	16	.9593	7.99	18,840	150,530	142,300	335.91	59.83
	5	18	.9465	7.89	18,930	149,360	140,900	331.42	59.03
4	5	20	.9340	7.78	19,020	147,980	139,600	327.05	58.25
4	5	22	.9218	7.68	19,110	146,760	138,300	322.81	57.49
4	5	24	.9100	7.58	19,190	145,460	137,100	318.65	56.75
4	2	26	.8984	7.49	19,270	144,330	135,800	314.58	56.03
4	2	28	.8871	7.39	19,350	142,990	134,600	310.63	55.32
	2	30	.8762	7.30	19,420	141,770	133,300	306.81	54.64
	2	32	.8654	7.21	19,490	140,520	132,100	303.03	53.97
	2	34	.8550	7.12	19,560	139,270	130,900	299.37	53.32
1	2	36	.8448	7.04	19,620	138,120	129,700	295.80	52.68
1	2	38	.8348	6.96	19,680	136,970	128,500	292.32	52.06
1	1	40	.8251	6.87	19,750	135,680	127,300	288.91	51.46
1	1	42	.8156	6.79	19,810	134,510	126,200	285.55	50.86

The above relation between specific gravity and A.P.I. degrees is expressed by the formula $\frac{141.5}{131.5^\circ + \text{A.P.I.}^\circ} = \text{Sp. Gr. at } 60^\circ\text{F.}$

For each 10°F. above 60°F. add 0.7° A.P.I.

For each 10°F. below 60°F. subtract 0.7° A.P.I.

Courtesy of Hauck Manufacturing Company



Courtesy of Power Magazine

Figure 10 — Probable ranges of BMEP during next ten years.

cost due to high wear offset the major cost advantage of the cheaper fuel. As a result, much development work has been done to determine the mechanism of this wear. It has been learned that a large measure of the wear rate results from corrosive attack of sulphurous combustion products. Increased engine cylinder temperature aids in reducing this attack, but in many large engines the high temperatures necessary cannot be tolerated. Work related herein outlines some of the results obtained in the design and application of cylinder lubricants for the prevention of this corrosive wear.

DEVELOPMENT OF NEW CYLINDER LUBRICANTS

In an attempt to reduce liner and ring wear when heavy fuels were used, many different qualities of lubricants have been tried in diesel engines on motorships, ranging from straight mineral oils to very high detergent type oils. Little if any change in wear was recorded. This work indicated that a cylinder lubricant, to successfully lubricate a large slow speed diesel, must incorporate characteristics to fulfill the following functions:

1. Provide neutralizing capacity to resist the corrosive attack of sulfurous combustion products in cylinder and ring metals.
2. Demonstrate extreme pressure properties.
3. Spread over cylinder surfaces.
4. Lubricant combustion products or lubricant itself should not be injurious to the engine metals.

One of the early products developed was an *emulsified oil* containing approximately 20 per cent water with calcium salts dissolved in the aqueous phase. Testing showed that this product effectively reduced cylinder wear and it is widely use for this purpose. However, as is the case with most emulsions, this product was relatively unstable, i.e., it tended to separate. Separation of components under the many conditions of storage and handling continues to be a serious problem.

A second non-conventional type of lubricant was next developed which consisted of a *dispersion* of calcium salts in the lubricating oil itself. This product also is effective in wear reduction and is widely used. Being a dispersion it also suffered from the problem of separation in storage and handling.

These conditions suggested the desirability of developing a homogeneous lubricant which would avoid unusual storage and handling problems yet incorporate the corrosion-neutralizing characteristics that the emulsion and dispersion types demonstrated.

Many lubricants have been developed and subjected to laboratory engine tests. One found particularly suited for the application contained a newly developed additive which was wholly soluble in lubricating oil. This product referred to herein as Experimental Oil X was subjected to field testing.

The field test results are from a diesel engine installed in power plant service operating 24 hours per day. The engine is a two-cycle, five-cylinder 21½" bore model with trunk type pistons. Each cylinder is equipped with an individual six-barrel lubricator as illustrated in Figure 7 which enables the use of different oils in the various cylinders during each test period. This permitted a direct comparison of cylinder lubricating oil performance in the same engine during the same test interval — which is most necessary for a true comparison inasmuch as fuel quality, engine operation and other factors can and do vary from one test period to the next in commercial operation.

The quality of the fuels used is demonstrated by the analysis of five samples taken at various times from storage. These analyses are shown in Table III. These fuels are of the commercial Bunker C type with Saybolt Furol viscosities at 122°F. ranging from 162 to 197 seconds, carbon residues of 8 to

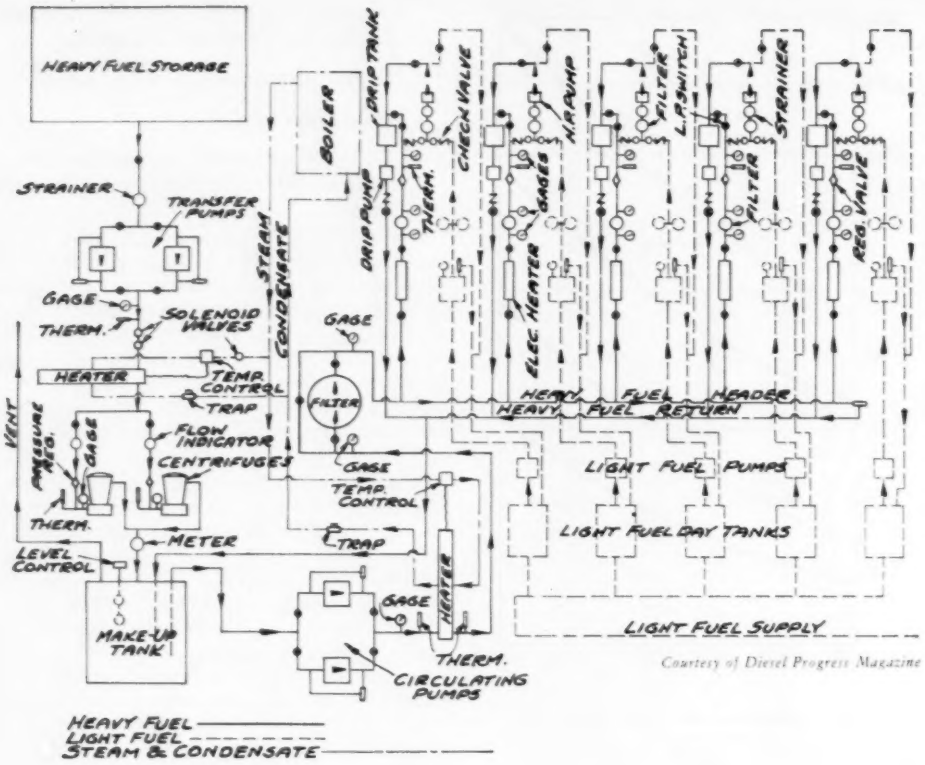


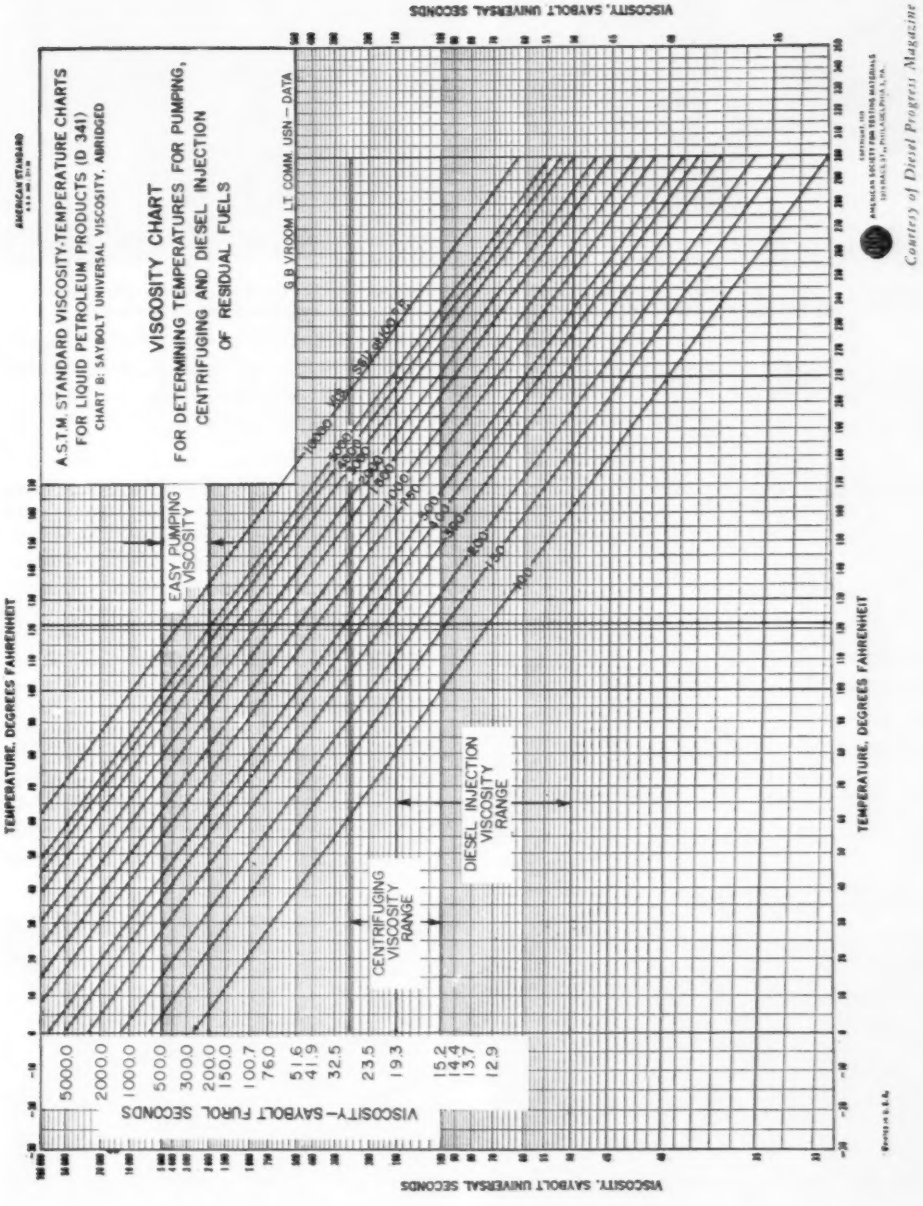
Figure 11 — Schematic Diagram of typical residual-fuel processing system.

17 per cent, and sulfur content from 1½ to 2½ per cent. It is particularly important to note that the gravities varied from 9.8 to 12.9° API with the majority of fuel consumed in the range of 10° API which is the gravity of water. Since fuels of this type normally contain some water contamination, the close correspondence between the gravities of the fuel and water makes removal of the latter by centrifuging very difficult. This situation points up the problem of fuel conditioning and the necessity of incorporating adequate heating, centrifuging and filtering facilities to prepare the fuel for delivery in a clean condition to the engine.

In Table IV are shown the physical tests on two heavy duty oils and Experimental Oil X. Heavy Duty Oil A and Heavy Duty Oil B are conventional MIL-L-2104A type lubricants. Experimental Oil X while different in composition was specifically formulated to combat corrosive wear from combustion products.

Experimental Oil X was evaluated during three test periods. Cylinder liner wear data are shown in Table V. It is noted that in Run D of 595 hours duration Experimental Oil X showed wear reduction of 37 to 72 per cent. In Run E of 1064 hours its wear reduction was 80 per cent and in Run F of 1450 hours its wear reduction was 50 to 64 per cent. It is further to be noted from a vertical comparison in any column that the liner wear in the same cylinder with the same lubricant varies from test period to test period. These variations undoubtedly occur largely because of changes in the quality and condition of the fuel delivered to the engine and variable engine loads. It is considered that the most valid relative wear performance is gained by this comparison of relative wear occurring during each test period.

It was determined that with the reduction of liner metal removal with Experimental Oil X there was also an accompanying reduction in ring wear. During the 1450 hour test period, loss of metal on the radial ring dimension was checked and is shown in Table VI. The reduction in radial wear is striking, particularly for the rings with higher wear. For com-



Courtesy of Diesel Progress Magazine

Figure 12 – Viscosity-temperature chart for determining temperatures for pumping, centrifuging and diesel injection of fuels.

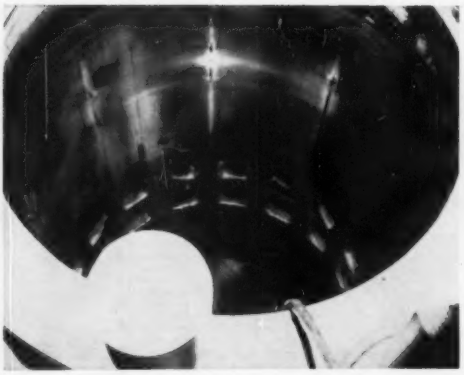


Figure 13 — Intake port deposits (Note five cylinder-lubricant injection holes).

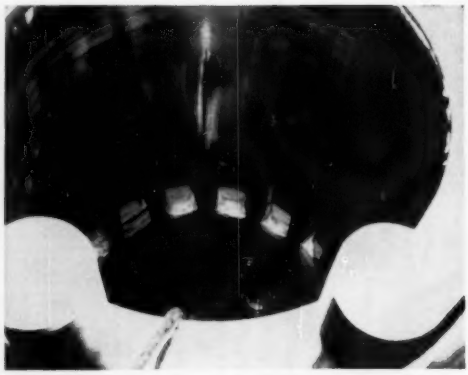


Figure 14 — Exhaust port deposits (Note three cylinder-lubricant injection holes).

pression ring 1 and 2 the reduction in wear rate is approximately 75 per cent.

The effect of lubricant on liner wear is shown graphically in Figure 18. This graph shows wear as the increase in maximum cylinder dimension for each of three cylinders operating on different oils.

TABLE III

Analyses of Fuel Oil Samples

Sample No.	56–8427	57–263	57–1878	57–2562	57–2563
Gravity, °API	12.9	9.9	9.8	11.0	9.9*
Water, %	Trace	0.1	0.3	0.4	3.2
Flash, PM, °F.	178	170	194	180	216*
Pour, °F.	25	50	50	35	45
Viscosity SUF at 122°F. Sec.	167	197	186	195	162*
Carbon Residue, %	8.0	17	14.9*	14.9*	17.0*
Ash, %	0.03	0.06	0.06	0.02	0.9*
Spectrographic Analysis, PPM					
Calcium	40	24	19	80	113
Iron	19	23	22	24	130
Nickel	16	18	16	16	28
Silicon	—	60	100	100	400
Sodium	—	33	56	24	338
Vanadium	12	27	34	20	27
Sediment by Extraction					
Percent	0.01	0.06	0.07	0.04	1.17
Sulfur, %	1.48	2.27	2.26	2.48	1.83
Hydrogen, %	10.7	9.98	10.3	10.5	10.0

*On Water-Free Sample.

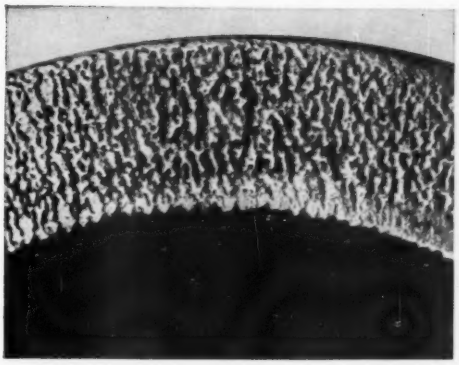


Figure 15—Corrosion of diesel engine valve head.

During the first 1223 hours cylinders 1 and 2 were operating on two different but conventional heavy duty oils and cylinder 4 on Oil A. During the next 1028 hour period all three cylinders operated on Oil A. During the following three test periods Heavy Duty Oil B was used in Cylinders 1 and 2 and Experimental Oil X was used in Cylinder 4. The striking leveling-off of the wear rate with Experimental Oil X as compared to the other three lubricants is clearly demonstrated.

In the utilization of some of the special types of cylinder oils in marine diesel service, rapid increases in engine bearing metal corrosion tendencies by the supposedly separate crankcase oil have been observed after relatively short periods of service. Analysis of these crankcase oils indicated they have been contaminated with the cylinder lubricant. During the period of testing, small quantities of the Experimental Oil X were found in the crankcase lubricant as is to be expected with a trunk type engine design. Frequent analysis of crankcase oil condition indicated no adverse effects.

TABLE IV

Physical Tests on Cylinder Lubricants

	Heavy Duty Oil A	Heavy Duty Oil B	Experimental Oil X
Flash, COC, °F.	460	460	500
Fire, COC, °F.	500	500	560
Gravity, °API	24.5	24.8	23.2
Viscosity, SSU			
at 100°F.	939	995	825
at 210°F.	75.9	81.7	78.5
Ash, %	0.31	0.58	2.9

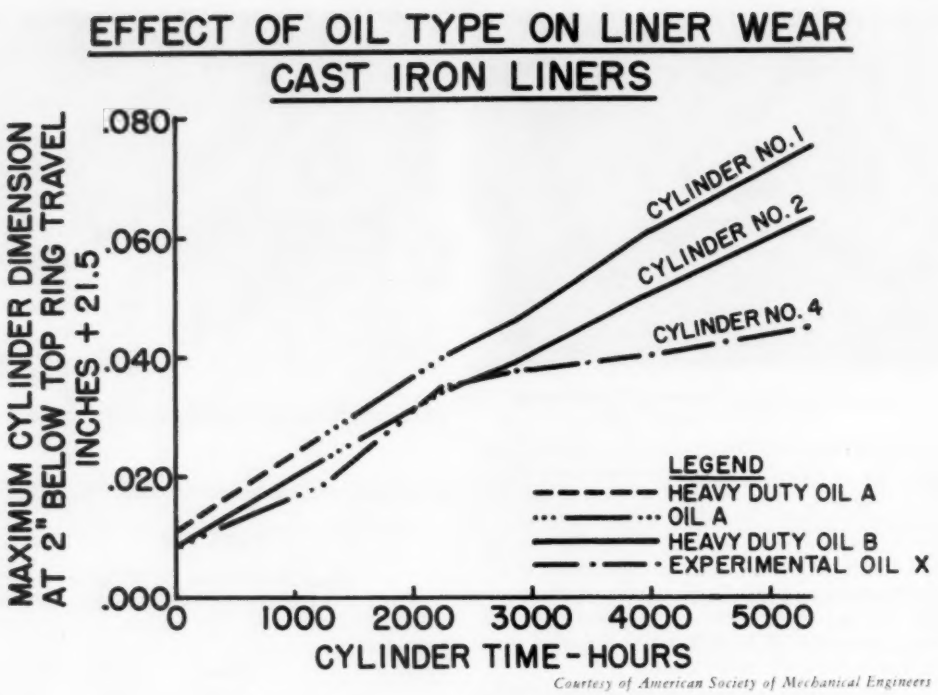
TABLE V

Effect of Oil Type on Wear of Cast Iron Cylinder Liner

Cylinder No.	1	2	4
Lubricant	Heavy Duty Oil B	Heavy Duty Oil B	Experimental Oil X
Run D (595 Hrs.) Thousandths in. Wear/1000 Hrs.	18	8	5
Run E (1064 Hrs.) Thousandths in. Wear/1000 Hrs.	15	15	3
Run F (1450 Hrs.) Thousandths in. Wear/1000 Hrs.	14	10	5

Figure 16 — Exhaust valve burning.

Figure 17—Magnified view of exhaust valve burning.



Courtesy of American Society of Mechanical Engineers

Figure 18 — Effect of oil type on cast iron liner wear.

TABLE VI

Effect of Oil Type on Piston Ring Wear
(Cast Iron Liners)

Maximum Radial Ring Wear — 1450 Hour Test
Thousandths in. Wear/1000 Hrs.

Lubricant	Heavy Duty Oil B	Experimental Oil X
No. 1 Compression Ring	202–259	56–56
No. 2 Compression Ring	117	12–33
No. 3 Compression Ring	57	8–17
No. 4 Compression Ring	12	6–6
No. 5 Compression Ring	15–16	8–11

SUMMARY

Trends indicate that diesel engines of increased horsepower obtained largely through supercharging and turbocharging will be available over a wide range of power. In the marine field, diesels are being built for ships of 100,000 tons displacement. With the high thermal efficiency of the low speed diesels as compared to marine steam turbine installations, the diesel will undoubtedly become an increasingly popular marine power plant. Improved lubricants have been developed that add substantially to ring and liner life in large low speed diesel engines operating on heavy fuels with high sulfur contents. New lubricants will make the diesel a more efficient consumer of the economical heavy fuel and will accelerate this trend to heavy fuel in both marine and stationary power plant service.

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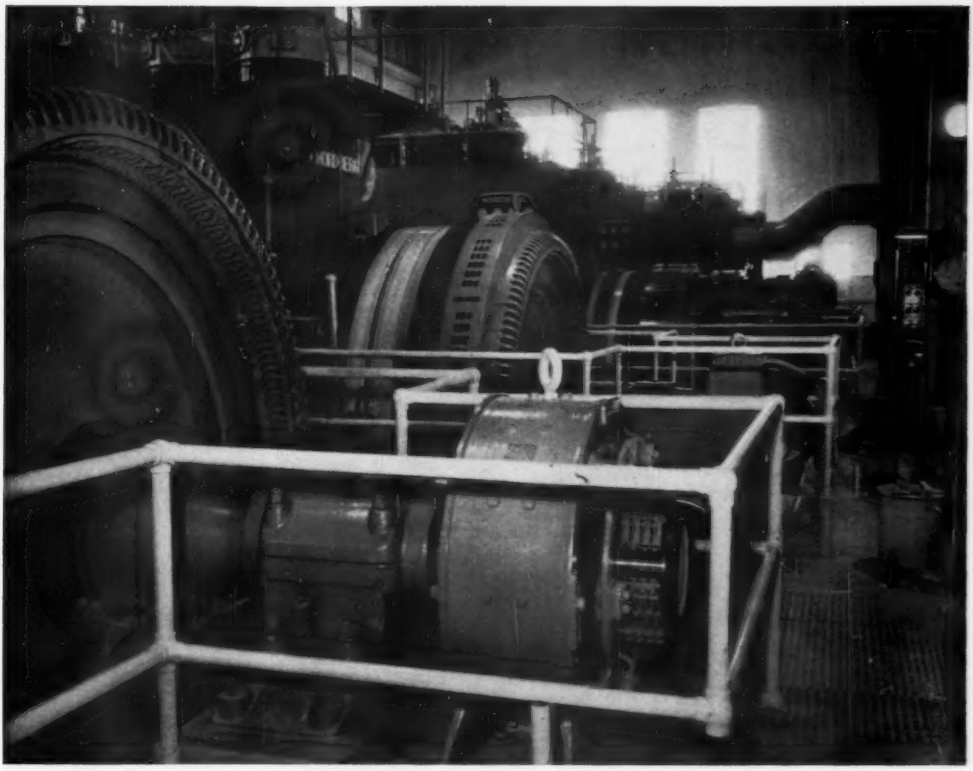

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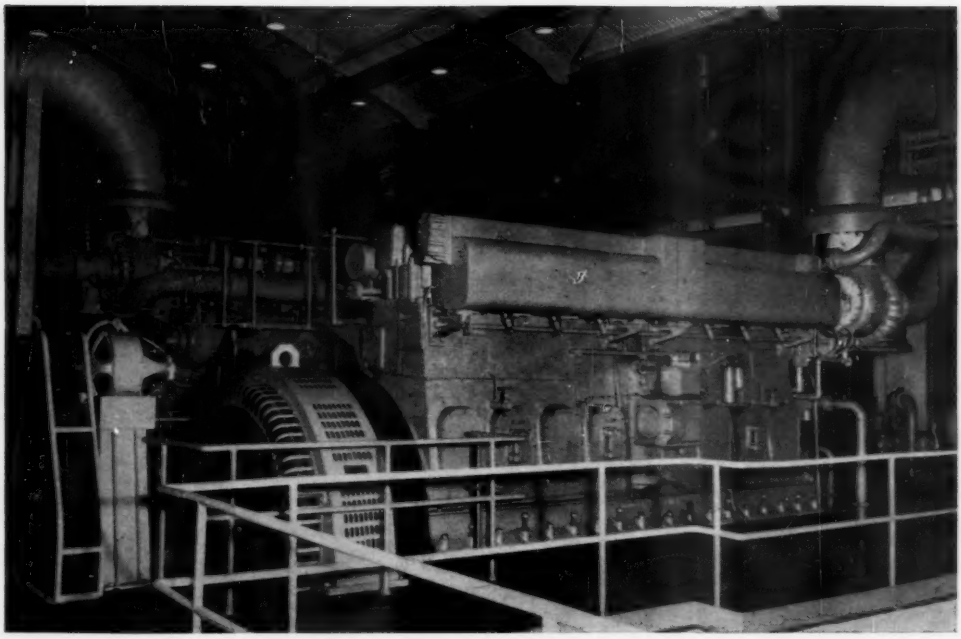

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